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Long-Term Strength of Steam-Turbine Rotors in the Stress Concentration Zone

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Abstract—The results from calculating the long-term strength of single-piece forged rotors using the natural mechanistic procedure [1] are compared with the experimental long-term strength characteristics of the models of high-temperature disk rims [2, 3] and with the field data on failures of such rims [4]. A new experimentally substantiated calculation procedure is proposed. The use of the procedure for estimating the long-term strength of the rotors of turbines now in service shows that the failure probability of the rotor rims of 800 MW turbines that have been in service for 100000 h is approximately equal to 30%.

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The term *natural mechanistic calculation procedure* is understood to mean a procedure based on universally accepted ideas. This procedure, the principles of which were described in [1], consists in brief in the following.

We consider the rotor deformation process that begins with initial an steady state, in which the rotor speed, steam temperature, and pressure have reached their steady rated values. After that, the parameters are maintained constant all the time until the service life is over. Once initially loaded, the entire rotor is usually in an elastic state. An assessment shows, however, that in some cases elastoplastic deformations of the material may occur in zones where considerable concentrations of stresses appear, and the actual stresses at the points of their concentration turn out to be lower than the values predicted on the assumption that the material is initially in an elastically strained state. Figure 1a shows a fragment of the intermediate-pressure rotor of a K-800-240 turbine. Figure 2 shows the change with time in the maximum normal stress $\sigma_\varphi(t)$ in the stress concentration zone in the rim fillet depicted in Fig. 1b, calculated using the natural procedure.

From a dependence like that shown in Fig. 2, we determine the equivalent design stress using the commonly accepted expression

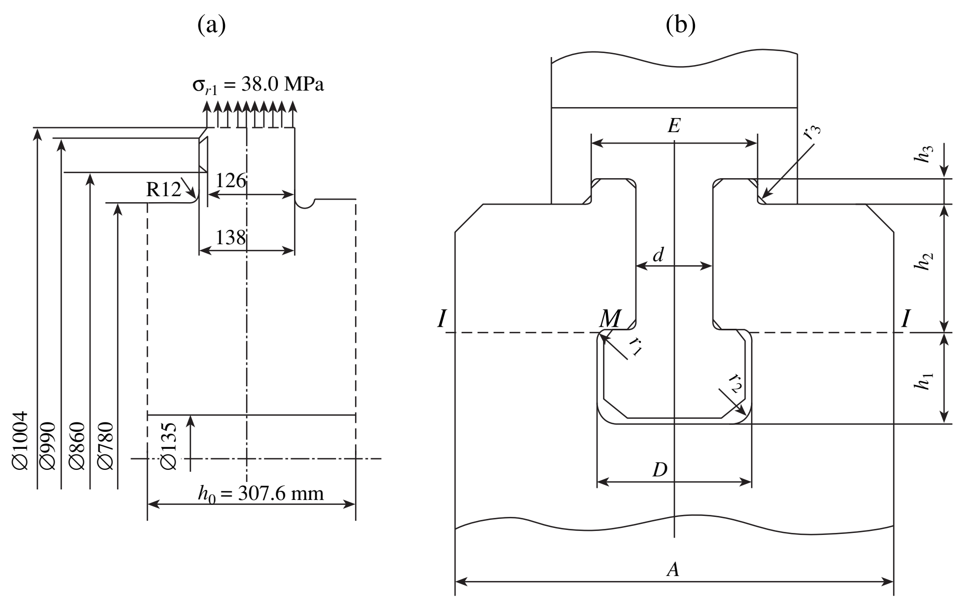

Fig. 1. Fragment of the intermediate-pressure rotor of a 800-MW turbine. (a) Design dimensions of the fragment of the intermediate-pressure rotor; (b) dimensions of the disk rim.

$$\sigma_{eq}^{f} = \left(\frac{1}{\tau}\int_{0}^{\tau}[\sigma_{\varphi}(0)]^{b}dt\right)^{\frac{1}{b}}, \quad (1)$$

where τ is the design service life, b is the exponent in the law describing the long-term strength of the material

$$t_f = B\sigma^{-b}; \quad (2)$$

t_f is the time to failure, σ is the stress, and B is the second constant in (2). Both constants are functions of temperature.

The following equivalent stresses have been obtained for the example considered (see Fig. 2):

when $\tau = 10^5$ h, $\sigma_{eq}^{10^5} = 239$ MP, and when $\tau = 2 \times 10^5$ h, $\sigma_{eq}^{2\times 10^5} = 228$ MPa.

According to the natural procedure for strength estimation, these equivalent stresses are compared with the ultimate long-term strength $\sigma_{l.t.}$ and the safety factors are determined as follows:

$$n_{l.t.} = \sigma_{l.t.}/\sigma_{eq}^{f}. \quad (3)$$

According to [5], at a temperature of 525°C $\sigma_{l.t.}^{10^5} = 181$ MPa and $\sigma_{l.t.}^{2\times 10^5} = 167$ MPa. It follows from this that, even for the service life $\tau = 10^5$ h, the coefficient $n_{l.t.} = 181/239 = 0.757$; i.e., it is less than unity, which means that the rims of all the turbines of this type should have already failed. But since this is not the case in reality, we should find out what the shortcoming of the natural procedure is and what corrections should be made to it.

Clearly, the procedure under discussion is rather conservative; hence, the choice made using this procedure at the design stage *will be reliable*, ensuring that the turbine will not suffer from severe accidents, which are usually caused by a failure of its rotor.

However, since a great number of turbines are now in service, the assessment of which using the above procedure shows them to be unreliable, we are facing the serious and urgent problem of introducing fairly justified refinements into this procedure for *making it possible to reliably predict the future of turbines that are in operation.*

We should note that the results of study [1], in which the reliability of turbines was estimated in three critical zones (the central channel, the disk fillet, and the relief holes in the disks), have shown the turbines to have sufficient strength margins for a service life of as long as 2×10^5 h, it is only the disk rim fillet (the fourth critical zone) that gives rise to anxiety.

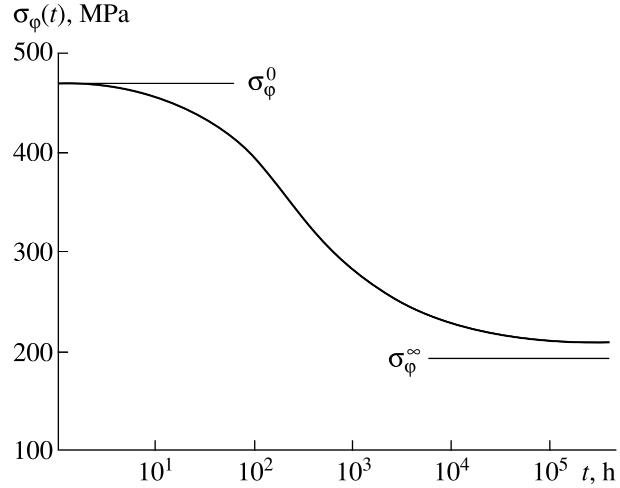

Fig. 2. Law for the change in the maximum tensile stress in the fillet of the disk rim in the first stage of an intermediate-pressure rotor of a 800-MW turbine, calculated using creep hypothesis.

An equivalent coefficient of stress concentration in the disk rim fillet is then introduced,

$$k_{eq}^{f} = \sigma_{eq}^{f}/\sigma_{rn}, \quad (4)$$

where $\sigma_{r.n}$ is the nominal stress in the disk rim wall (in section *I–I*, Fig. 1b).

Coefficient (4) gives a design estimate of the effect of stress concentration in the fillet zone on the long-term strength of the rim.

EXPERIMENTAL INVESTIGATIONS OF MODELS OF DISK RIMS WITH T-SHAPED ROOT JOINTS OF BLADES [2, 3]

If we wish to obtain a realistic estimate of the long-term strength of the disk rims of turbines that are in operation, it is best to use the results of comprehensive investigations of rim models reported in [2, 3]. The main results of these investigations are given in Table 2 of [3].

Figure 3 shows the results obtained from processing the primary data of [3] using the least-squares method. Curves *1* and *2* show the nominal stresses in the rim walls (see Fig. 1b, section *I–I*),

$$\sigma_n = P/F\,[F = (A - D)b_0], \quad (5)$$

as a function of the time to failure of the models, where P is the total force applied to section *I–I* by the loading device of the tearing machine. The force in section *I–I* was transferred through the blade root model. Curve *1* is for the models of standard size 1, and curve *2* is for the models of standard size 2. The dimensions of the models are given in Table 1. The configuration of the rim model profiles is quite similar to that of the actual rim (Fig. 2b) The dimensions of the actual rim profile

Table 1. Basic geometric parameters of the profiles of the models, mm (the thickness of the models b_0 = 20 mm)

Standard size	A	D	d	E	h_1	h_2	h_3	r_1	r_2	r_3	$\alpha_{\sigma\varphi}$
1	120	54	30	56	26.5	25	7	3	7	2	9.5
2	92	38	19	36	22.5	27	5	2	8	1.2	10.0
3	126	45	22	–	27	36	–	2.5	–	–	11.7

Table 2. Parameters of dependence (2)

Standard size	B'	b'	B''	b''	σ_n^{infl}, MPa	t_f^{infl}, h
1	9.1425×10^{23}	9.9299	5.9661×10^{11}	3.8362	99.93	12718
2	9.2504×10^{22}	9.4644	8.9243×10^{10}	3.2926	94.56	18511

Note: The constants for the samples (curve *3* in Fig. 3): $B = 1.90385 \times 10^{32}$; $b = 12.0612$.

are given in line 3 of Table 1; the theoretical values of the coefficient for stress concentrations in the zone M on the fillet surface found from [5] are given in the last column of the table.

Curve 3 in Fig. 3 shows the long-term strength of the cylindrical samples made of the material (steel 25Kh1M1F–R2MA) obtained from the same heat as that of the rim models as a function of t_f. The models and samples were tested at a temperature of 540°C.

The parameters of dependence (2), which was taken as the approximating one, for different sections of the models are given in Table 2. The quantities marked with a prime are for the first sections before the inflection point of the strength curve, and those marked with a double prime are for the sections after the inflection point.

Table 2 also gives the coordinates of the inflection points on the long-term strength curves. According to these estimates, which should be regarded as approximate ones since the number of test points is rather small, the conventional inflection for standard size 1 is at around 13000 h, and that for standard size 2 at around 19000 h of the test.

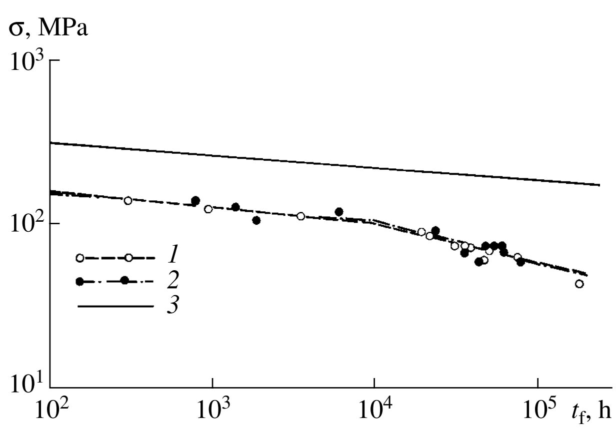

Fig. 3. Long-term strength of the models and cylindrical samples made of steel R2MA of the same heat [2, 3]. (*1*) Model of standard size 1, (*2*) model of standard size 2, and (*3*) cylindrical samples.

Table 3 gives the results obtained from the tests of the cylindrical samples (line 1) and models of the rims (lines 2 and 3) in accordance with the parameters given in Table 1. Here, σ_{n1} and σ_{n2} are the nominal stresses in models 1 and 2, respectively.

Lines 4 and 5 of Table 3 contain the values of the quantity

$$k_{eq}^{exp} = \sigma_{l.t.}/\sigma_n, \tag{6}$$

which can be termed *the experimental equivalent stress concentration factor* in the rim fillet, which *determines the reduction in the long-term strength of the rims compared with the cylindrical samples.*

The dependences of σ_{n1} and σ_{n2} as functions of the time to failure (see curves *1* and *2* in Fig. 3) differ considerably from the dependence of $\sigma_{l.t.}$ for the samples (curve *3*) not only in the noticeable influence of stress concentrations, but also in their nature. The dependences for the models (curves *1* and *2*) have a noticeably varying slope (inflection). The change in the slope is associated here with a change in the kind of failure [2, 3]. When the models collapse in the first sections of the curves (before the inflection), their failure is ductile in nature, with considerable plastic deformation before the failure. When the models collapse in the second sections (after the inflection on curves *1* and *2*), their failure is usually accompanied by a crack that occurs in the fillet zone. This crack develops without considerable plastic deformation; i.e., the collapse is close to a brittle failure in nature.

As a rule, *this is connected with the transition from a transgranular fracture to an intergranular fracture.* Hence, *an increased local stress in the fillet* (due to stress concentration) *contributes to embrittlement of the material.*

Table 3. Consolidated data on the long-term strength of the samples and models (experimental and calculated)

No.	Parameter	t_f, h					
		10^3	10^4	2×10^4	5×10^4	10^5	2×10^5
1	$\sigma_{l.t.}$, MPa	267.7	221.1	208.8	193.5	182.7	172.5
2	σ_{n1}, MPa	129.1	102.4	88.8	69.9	58.4	48.7
3	σ_{n2}, MPa	128.7	100.9	92.4	69.9	56.7	45.9
4	k_{eq1}^{exp}	2.074	2.159	2.351	2.768	3.128	3.542
5	k_{eq2}^{exp}	2.080	2.191	2.226	2.747	3.783	3.758
6	k_{eq1}^{f}	3.821	3.807	3.890	4.064	4.226	4.402
7	k_{eq2}^{f}	3.884	3.929	3.896	4.177	4.416	4.623
8	q_1	0.381	0.413	0.467	0.577	0.652	0.747
9	q_2	0.374	0.407	0.433	0.557	0.650	0.761

Note: Subscripts 1 and 2 denote the standard sizes of the models.

Accelerated embrittlement in the stress concentration zone makes it much more difficult to extrapolate data on long-term strength obtained in tests with short periods (about 10^4 h) for the actual service lives (from 10^5 to 2×10^5 h) or longer.

CALCULATING THE EXPERIMENTAL RIM MODELS FOR LONG-TERM STRENGTH USING THE NATURAL PROCEDURE

The problem is formulated as follows. Given the conditions under which experiments are carried out on the models, we calculate k_{eq}^{f} using (4) and compare its values with the experimental data in Table 3. The natural procedure will thereby be directly compared with the experiment.

The results obtained from calculating k_{eq}^{f} for the conditions under which the models were tested are also given in Table 3. Comparing the experimental values of k_{eq}^{f} (Table 3, lines 4 and 5) with their predicted values (lines 6 and 7), we see that they differ from one another, the difference being especially large for short times to failure ($t_f < 10^5$ h). It should be pointed out, however, that the distant extrapolation is far from being well justified.

THE COEFFICIENT FOR THE SENSITIVITY OF MATERIAL TO STRESS CONCENTRATION

To bring the results of calculations into agreement with the experimental data, we introduce *the coefficient q for the sensitivity of the material* to stress concentration under creep conditions,

$$q = (k_{eq} - 1)/(k_{eq}^{f} - 1), \tag{7}$$

where k_{eq} is *the true coefficient of stress concentration* through which we determine the equivalent stress,

$$\sigma_{eq} = k_{eq}\sigma_n; \tag{8}$$

and σ_n is the nominal stress for the concentration zone.

The value of k_{eq}^{f} has been obtained using the natural procedure and is determined from (4).

The coefficient q has been introduced by analogy with the method used to calculate parts for high-cycle fatigue, where it takes into account the reduction in the actual (effective) coefficient of stress concentration under cyclic loading, compared with the theoretical coefficient of stress concentration for a single elastic straining.

Using experimental data and results from calculating k_{eq}^{f}, we can find the values of q using (7) for the case when the models collapse. To do this, we take the experimental values of k_{eq}^{exp} as k_{eq}. The values of q are given in Table 3 and shown in Fig. 4 as functions of t_f. A characteristic feature of the change in $q(t_f)$ is that the value of q is approximately constant until $t_f \approx 10^4$ h: $q \approx 0.4$. This is a region where essential plastic deformation is observed when the models are initially loaded in the stress concentration zone. When $t_f > 10^4$, *the fracture becomes more and more brittle in nature*; we observe q growing first slowly and then more and more considerably, reaching 0.65 at $t_f = 10^5$ and 0.76 at $t_f = 2 \times 10^5$ h. The dependence of q on t_f can by no means be regarded as a physical one. It is rather a phenomenological fact, and the internal physical dependence of q on certain parameters is hidden, as only a limited amount of experimental data is available. Most probably, the influence of t_f on q reflects the effect of the exposure to high temperatures *and high local stresses*. When loading is

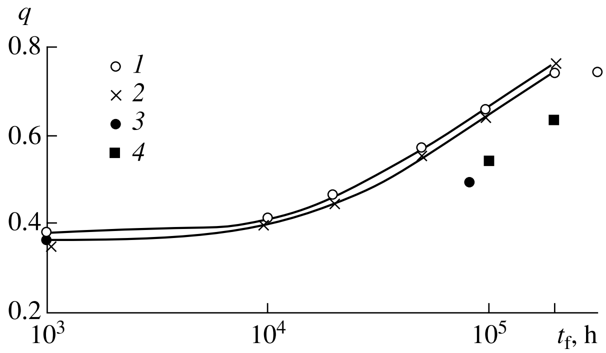

Fig. 4. Sensitivity coefficient q with t_f used as the governing parameter. (*1*) Model of standard size 1, (*2*) model of standard size 2, (*3*) experimental point for the rim of the intermediate-pressure rotor of a 500-MW turbine, and (*4*) recalculation of the data for standard size 2 for 525°C.

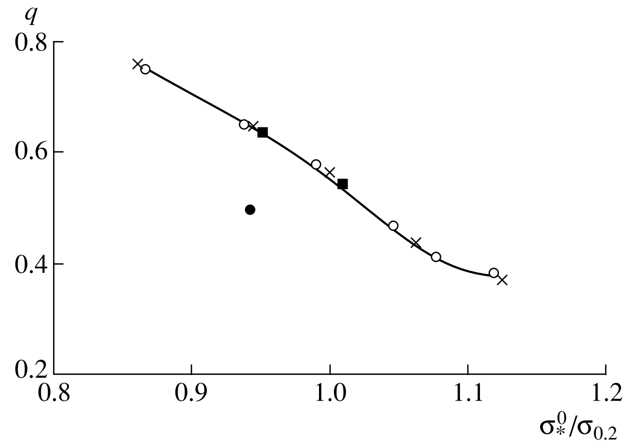

Fig. 5. Sensitivity coefficient q with $\sigma_*^0/\sigma_{0.2}$ used as the governing parameter. The notation is the same as in Fig. 4.

applied, the local stresses reach the highest values for the entire period of creep.

Suppose the influence of initial stresses to be a significant factor. Figure 5 depicts q as a function of the initial stress intensity σ_*^0 during the initial loading of the rim model. We can see from the figure that the curves for the models of standard sizes 1 and 2 virtually coincide with one another. According to the graph in Fig. 5, the effect of the difference in the stress concentration coefficients is taken into account by σ_*^0, the value of which is different for different concentrators. Thus, we have grounds to consider σ_*^0 a significant parameter for q, and the dependence $q(\sigma_*^0)$ can be taken as a universal one for the R2MA material. The limited amount of experimental data does not allow us to give preference to one of the methods for selecting the key parameter significant for q.

TECHNIQUE FOR CALCULATING THE LONG-TERM STRENGTH OF A DISK RIM USING THE TESTS ON THE MODELS

This technique is based on the natural procedure for calculating the equivalent creep stress with introduction of the sensitivity coefficient q to take into account the peaking nature of stresses at a critical point.

The actual equivalent stress is found from (8), where, according to (7),

$$k_{eq} = 1 + q(k_{eq}^f - 1). \tag{9}$$

The value of k_{eq}^f is calculated from (4) using the natural procedure, and q is found from one of the curves shown in Figs. 4 and 5 depending on the chosen governing parameter: t_f or $\sigma_*^0/\sigma_{0.2}$.

The value of σ_{eq} is compared with that of $\sigma_{l.t.}(t_f)$. If $\sigma_{eq} = \sigma_{l.t.}$, this means that the rim will fail at certain t_f (e.g., 10^5 h) with a certain probability (e.g., 50%).

The conventional safety factor for long-term strength is determined, as is usually done, using the expression

$$n_{l.t.} = \sigma_{l.t.}/\sigma_{eq}. \tag{10}$$

If we take $\sigma_{l.t} = \sigma_{l.t}(0.99)$ in (10), the acceptable values will be $n_{l.t.} \geq 1$.

EXAMPLE OF USING THE METHOD TO CALCULATE THE FIRST-DISK RIM FOR THE INTERMEDIATE-PRESSURE ROTOR OF A K-800-240 TURBINE

The calculation is carried out using the dependence $\sigma_\varphi(t)$ in the zone M (see Fig. 1b), shown in Fig. 2 and calculated using the natural procedure [1]. The characteristics of steel R2MA at a temperature of 525°C, taken from different sources ([6] and others), were used as initial data.

The initial maximum stress σ_φ^0 in the fillet, calculated in accordance with [1] taking the elastoplastic deformation at the critical point into account, was found to be 470 MPa. The ultimate steady-creep stress σ_φ^∞ = 192.8 MPa is shown in Fig. 2.

The ultimate long-term strengths of steel R2MA at 525°C are given in Table 4, where P, % is the probability of the material not failing; $\sigma_{l.t.}$, MPa is the long-term strength at different P and t_f; and t_f is the time to failure.

For the example considered, we have found $\sigma_{eq}^f(10^5)$ = 239 MPa according to [1] and k_{eq}^f = 5.25 using (4).

From (9), we can obtain

$$k_{eq} = 1 + q(k_{eq}^f - 1) = 1 + 4.25q, \tag{11}$$

and then from (8) we have

$$\sigma_{eq} = 5.5(1 + 4.25q), \qquad (12)$$

where q is found from the curves in Figs. 4 and 5 depending on the governing parameter. In our case the governing parameters are $t_f = 10^5$ h and $\sigma_*^0/\sigma_{0.2} = 0.93$. The values of q are found from Figs. 4 and 5, and from them we determine σ_{eq} using (12). The results are given in Table 5.

In this particular case, the choice of the governing parameter has little effect on the equivalent stress value.

Comparing the data of Tables 4 and 5, we find that the safety factor for the long-term strength of the rim, determined for an operation time equal to 10^5 h and a failure probability of 50%, is

$$n_{l.t.} = \sigma_{l.t.}/\sigma_{eq} = 181/173 = 1.05, \qquad (13)$$

i.e., slightly greater than unity. This means that the failure probability of the rim is slightly lower than 50%. Assuming (as is frequently done) that the mean standard deviation of $\sigma_{l.t.}$ is ±10%, then at $n_{l.t.} = 1.05$ we can obtain the rim failure probability $P_f \approx 30\%$. This means that *three out of ten turbines that have been in operation for 10^5 h under rated operating conditions may suffer from rim failure*.

THE EFFECT OF TEMPERATURE ON THE SENSITIVITY COEFFICIENT

The sensitivity coefficients q that have been introduced and calculated (Figs. 4 and 5) are, strictly speaking, for the 540°C temperature at which the rim models were tested. This is because the dependences $\log\sigma_n - \log t_f$ have inflections (see Fig. 3). Additional assumptions must be made if we wish to recalculate the characteristics (curves *1* and *2* in Fig. 3) for other temperatures. These assumptions can have a considerable effect on the results of the recalculation and, consequently, on the long-term strength of the rims being analyzed. If we assume that the broken curves $\log\sigma_n - \log t_f$ can be recalculated for the working temperature (525°C) using the Larsen–Miller method, the inflection points will shift to the region of larger t_f, and this will result in higher destructive stresses at the extrapolation values of $t_f = 10^5$ and $t_f = 2 \times 10^5$ h.

As an example, Figs. 4 and 5 show the points corresponding to the recalculation of the characteristics of the models (curves *1* and *2* in Fig. 3) for a temperature of 525°C using the Larsen–Miller method. It is interesting that these points lie precisely on the dependence $q(\sigma_*^0/\sigma_{0.2})$ (see Fig. 5); however, when calculated using the dependence shown in Fig. 4, the values of q obtained from these points differ considerably from those for a temperature of 540°C. In other words, the calculated strength of the rims depends considerably on the governing parameter adopted. If we take $\sigma_*^0/\sigma_{0.2}$ as the governing parameter, we obtain the estimate of the rim failure probability given in the calculation example (30%). If we take t_f as the governing parameter (see Fig. 4), we obtain a lower value of q and, hence, a higher long-term strength of the rim. It should be emphasized that, since an experimental justification of the possibility of making an optimistic assumption is lacking, there is nothing to do but content ourselves with the conservative estimate obtained in accordance with the proposed procedure and using the sensitivity coefficient q as shown in Fig. 5.

Table 4. Characteristics of steel R2MA at 525°C

P, %	t_f, h	$\sigma_{l.t.}$, MPa
50	10^5	181
50	2×10^5	167
99	10^5	171
99	2×10^5	159

Table 5. Equivalent stresses estimated taking into account the sensitivity coefficient q

Graph	q	k_{eq}	$\sigma_{eq}^{10^5}$, MPa
Fig. 4	0.65	3.76	171
Fig. 5	0.66	3.81	173

The calculation example carried out using the proposed procedure has shown that the long-term strength of the rims of 800 MW turbines is insufficient even for a service life of 10^5 h. In such a situation, it would be reasonable that the disk rims of turbines that are in operation be checked at regular intervals for cracks in their fillets.

It is noteworthy is that a few K-800-240 turbines now in operation are equipped with a system for cooling the intermediate-pressure rotor in the zone of the first stages [7]. According to design calculations, this system (which has been developed at the TsKTI) reduces the rim temperature to 480°C. The use of the cooling system appears to solve the problem of ensuring the long-term strength of the disk rims on the intermediate-pressure rotors of the K-800-240 turbines for as long as 200000 h.

CONCLUSIONS

(1) The long-term strength of the disk rims of steam turbines is in reality higher than that predicted using the conservative procedure [1].

(2) The procedure has been refined using the experimental data on the long-term strength of the rim models [2, 3] and field data.

(3) The use of the proposed experimentally supported procedure for calculating the long-term strength of the intermediate-rotor disks of a K-800-240 turbine gives a rim failure probability of about 30% after 100000 h of operation. This means that *rim failure may occur in three of out of ten turbines that have been in operation for 100000 h*.

(4) According to some optimistic conjectures, the failure probability falls, but these conjectures lack experimental substantiation.

(5) If we wish to ensure the required reliability of the rims, it is recommended that:

(i) the fillets be regularly checked for the occurrence of cracks in them and

(ii) a steam cooling system of the intermediate-pressure rotor be used.

(6) Using the proposed procedure at the design stage will give optimally conservative results.

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